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Short Communication

Latest concepts for combustion and waste heat recovery systems being considered for hydrogen engines

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ARTICLE INFO

Article history:
Received 13 December 2012
Received in revised form
14 January 2013
Accepted 16 January 2013
Available online 8 February 2013

Keywords:
Combustion systems
Waste heat recovery
Gas engines

ABSTRACT

A more sustainable transportation calls for the use of alternative and renewable fuels, a further increase of the fuel energy conversion efficiency of internal combustion engines as well as the reduction of the thermal engine energy supply by recovering the braking energy. The paper presents two concepts being developed to improve the fuel conversion efficiency of internal combustion engines for transport applications. The first concept works on the combustion evolution to increase the amount of fuel energy transformed in piston work within the cylinder. The second concept works on the waste exhaust and coolant energies to be recovered through a power turbine downstream of the turbocharger turbine on the exhaust line and a steam turbine feed with the steam produced by a boiler/super heater made of the coolant passages and a heat exchanger on the exhaust line. The concepts work with hydrogen (and in this case a water injector is also necessary) as well as lower alkanes (methane, propane, butane). Preliminary simulations show improvement of top fuel conversion efficiencies to above 50% in the high power density operation. The waste heat recovery system also permits faster warm-up during cold start driving cycles.

1. Introduction

The internal combustion engines fuelled with traditional or alternative fuels in hybrid vehicles is the best mobility option available at least for the next 20 years. The hydrogen based transport may also be preceded by a methanol based transport in the medium term. The methanol society could be an interesting intermediate step if renewable energy should be made available in large quantities in the near future, but fossil fuels should still be used largely in the power generation. In this case the methanol would be a much better transportation fuel than hydrogen (an almost ‘drop-in’ alternative to gasoline) produced at no extra energy cost from a feed stock of hydrogen and carbon dioxide. The hydrogen engine coupled to a hybrid power train is competing with electric vehicles as the preferred post fossil fuels society. The contribution provides some indications of where to focus the R&D of hydrogen internal combustion engines where the R&D for hydrogen may be beneficial also to the R&D for other alternative fuels as liquefied petroleum gas (LPG), compressed natural gas (CNG) or liquefied natural gas (LNG) of most immediate interest.

Different solutions are presently explored to produce high power density, high top and part load efficiency internal combustion engines with target related to the increased level of complexity vs. a traditional gasoline or Diesel engine. The most promising couples the direct injection of the hydrogen with jet ignition or the direct injection of pilot Diesel and eventually of water. The fuel efficiency mark is about 50% for

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http://dx.doi.org/10.1016/j.ijhydene.2013.01.112

these solutions [5–7], to be further improved through a better recovery of the waste heat in the coolant, the exhaust and the charge cooler [9,10].

This short communication is not a survey of the combustion and waste heat recovery systems being developed for hydrogen engines, but only a report of the latest concepts being considered by the author's group.

2. Novel combustion systems

Hydrogen is a very challenging fuel for transport applications having many conflicting properties, from the positive very high heating value per unit mass, fast laminar flame speeds and wide lean combustion limits, to the low heating value per unit volume, the many abnormal combustion phenomena and the significant safety issues for the flammability. Hydrogen powered vehicles may use ICE to propel the vehicle using hydrogen fuel and oxidizer (air or oxygen), plus other fuels like diesel to control the start of combustion, plus other fluids like water to control the combustion evolution and the rates of heat release and heat transfer. Many different engine combustion concepts have been proposed, mostly of the premixed type, homogeneous or stratified, in some cases of the diffusion type, in the latest studies of mixed premixed-diffusion type.

The options we have considered are port fuel injection (PFI) with stoichiometric and lean combustion, direct injection (DI) with stoichiometric, lean and stratified combustion, premixed, diffusion and mixed modes of combustion with novel DI and ignition strategies, with spark ignition, jet ignition or diesel fuel ignition, with or without turbocharging or waste heat recovery, in 2 stroke and 4 stroke arrangements, with oxygen or air as the oxidizer, and with or without water injection.

The use of hydrogen PFI is considered in [1,2]. This engine is a traditional gasoline engine modified to inject hydrogen. A jet ignition device, a small pre-chamber connected to the main chamber through calibrated orifices and accommodating a second fuel injector for the hydrogen and a spark plug replaces the traditional main chamber spark plug to provide a much more efficient ignition event. Rather than a low energy near wall process, with jet ignition the main chamber premixed mixture is bulk ignited with high energy. The jet ignition permits in some extent to control the load of the engine by changing the amount of hydrogen injected in the port. The operation throttle-less, diesel like permits to retain similar to diesel efficiency penalties reducing the load because the throttling losses typical of the traditional gasoline engines are avoided. The load is reduced by reducing the quantity of fuel injected and not by throttling the intake. The blockage effects for the low density hydrogen injected in the port are solved for the specific case with cryogenic hydrogen also permitting operation with high boost ratio of the turbocharged engine and use of high compression ratio pistons. Because of the cryogenic injection, the engine has a significant power density and top efficiencies in the low 40% mark.

The requirement of cryogenic hydrogen injection may be removed by using direct injection. By introducing the hydrogen fuel directly in the combustion chamber, the low density of the hydrogen is not penalising the power density. This also significantly reduces the occurrence of abnormal combustion phenomena that have historically significantly limited the port injected hydrogen engine [19]. The direct injection introduces an additional advantage, the opportunity to burn stratified. While the PFI produces a mostly homogeneous fuel–air mixture to be ignited within the cylinder, the DI may in principle also permit to produce stratified fuel mixtures with combustion occurring much easier with overall very lean mixtures and reduced heat losses to the walls. The DI of hydrogen coupled to jet ignition is considered in [3,4]. The major issue of this design is the availability of hydrogen injectors permitting very high mass flow rates and multiple injections similarly to the latest Diesel injectors. This problem has been partially alleviated by the latest Westport hydrogen injectors like J43 or J44 permitting much higher flow rate than ever before sequentially [20]. The design has top efficiencies in the low-to-mid 40% and better operation especially at very low loads because of the stratification process.

The designs of [1–4] are basically of a premixed gasoline-like engine. The option of designing a diffusion diesel-like engine may also be of interest, because of the possible advantages of the diffusion combustion vs. the premixed combustion of hydrogen in terms of fuel conversion efficiency. The auto-ignition of hydrogen is much more difficult than the auto ignition of the Diesel, and some additional measure is needed to burn the hydrogen as soon as it diffuses in the air.

The dual fuel option where a pilot Diesel direct injection creates the condition for a subsequent directly injected gaseous fuel to burn diffusing in air Diesel-like is a very well established technique. This technique is explored in [5] where a double injector engine cylinder has the pilot direct injection totalling 5–10% of the total fuel energy intake creating the proper conditions for the subsequent direct injection of the hydrogen to burn controlled by diffusion diesel like. This dual fuel option permits about same of the Diesel fuel efficiencies full and part load with eventually the opportunity to increase the power density running lower air-to-fuel equivalence ratios for the quickest mixing with the air of the hydrogen vs. the complex liquid hydrocarbon mixture of the Diesel [5–7]. The design has top efficiencies in the mid 40% and better operation especially at very low loads because of the diffusion combustion process.

The opportunity to ignite a premixed mixture of hydrogen and air with a direct injection of a Diesel fuel or the jet ignition may be coupled to the opportunity to create suitable conditions for the subsequently injected hydrogen to burn diffusing in hot air by further expanding both the dual fuel concept, with a diesel direct injector and a hydrogen direct injector being accommodated in the cylinder head together with a glow plug, or by using a jet ignition device of different characteristics and the hydrogen direct injector. This opportunity is explored in [6,7]. The hydrogen injected before the igniting event burns premixed, the hydrogen injected after the igniting event burns diffusion, it is possible to have a gasoline-like combustion permitting high power densities as well as a diesel-like combustion permitting high fuel conversion efficiencies and obviously mixed modes modulating the injection parameters. This solution permits fuel efficiencies in the mid-to-high 40% with very high power density ratios.

Enablers of these novel ICE are novel gas injectors, permitting fast, high flow rate operation with the option of

multiple injection strategies, and the jet ignition devices, an idea some decades old still waiting for some support to perform a decent R&D without any major technological obstacle to achieve good results.

Apart from the combustion evolution premixed, diffusion or mixed, two other areas are of major concern for the development of hydrogen ICE, the water injection (WI) and the waste heat recovery (WHR).

WI is proposed in [8,9,11] to control the occurrence of knock in the premixed charge, the temperature of gases to turbine in turbocharged applications, to reduce the heat transferred to the walls and to increase the piston work by the steam expansion. Port water injection (PWI) is a well known technology to reduce the knock sensitivity and reduce the top temperatures within the cylinder and to turbine in case of turbocharged engines. Direct water injection (DWI) permits in addition to the above advantages the opportunity to more precisely act on the heat release and the heat transfer rates, and to boost the piston work by expanding the steam. Without water injection, an ICE may hardly operate close to stoichiometry because of the abnormal combustion phenomena, the significant heat losses and the extremely high temperatures.

WHR is proposed in [9,10] to further boost the thermal efficiency recovering the exhaust and coolant heat otherwise lost. Much more than 50% of the available fuel energy ultimately ends in the exhaust gases and the coolant waste heat. A significant amount of the higher temperature exhaust waste heat and a percentage smaller but still important amount of the coolant heat may be recovered by using a waste heat recovery (WHR) system of the Rankine type. The best option presently available is provided by the integration of the heat recovery system with the engine, having the coolant passages serving as boiler/pre-heater and then having a closed-coupled heat exchanger on the exhaust serving as boiler/super-heater, and water as the working fluid. This system has reduced weight and improved packaging vs. other WHR systems, permits better efficiencies and a much faster warm-up. The benefits are not only the improvement of full and part load fuel conversion efficiencies, but also a quicker warm-up reducing the fuel penalties during cold start because the exhaust and coolant energies are both used to accelerate the warm-up [10]. The use of oxygen rather than air as oxidizer is an option not explored so far in hydrogen ICE but only limited to fuel cells. Oxygen would be available for free if the hydrogen is produced by electrolysis of water. Combustion in oxygen does not produce nitrogen oxides, but the oxygen must also be stored on board with additional complications. The ICE may however operate much more efficiently being the combustion rate in oxygen much faster than in air, with the major downfall the much higher flame temperatures. If coupled with direct water injection, an ICE with hydrogen and oxygen direct injection and a downstream power turbine may achieve efficiencies in the mid to high 50% also adopting the higher power density two strokes rather than a conventional four stroke architecture [11].

It is worth of mention that the use of jet ignition or glow plug when combined with premixed combustion comes with a risk of the destructive combustion knock which is similar to spark ignition combustion knock. This risk reduces the amount of hydrogen that would burn premixed vs. the amount of hydrogen that would be diffusion, i.e. the amount of hydrogen injected before the igniting event vs. the amount of hydrogen injected after the igniting event is started. This is certainly a limiting factor in achieving greater gain in BMEP and efficiency to be addressed with proper calibration of the injection and ignition events. The water injection is not supposed to be used to control the occurrence of knock but control the temperature of gases for heat transfer purposes and does not occur prior of the start of combustion. Water injection used to mitigate the knock problem would otherwise affect the combustion efficiency and quality.

Fig. 1 presents the direct injection jet ignition combustion system concept. The combustion chamber is a bowl-in-piston, high compression ratio combustion chamber. The H_2 injector in the main chamber delivers multiple injections. Fig. 2 presents a sketch of the combustion chamber and the jet ignition pre-chamber of a direct injection spark ignition gasoline engine gasoline engine modified to run hydrogen direct injection and jet ignition (from [4]). The H_2 injected in the main chamber before the jet ignition event burns premixed. The H_2 injected after the jet ignition event burns diffusion. The jet ignition event is controlled by the spark discharge after the H_2 injector in the pre-chamber has created a stoichiometric mixture in the pre-chamber. The H_2O delivers multiple injections. Before combustion, vaporisation of the water reduces the tendency to knock. After combustion, occurring almost isochoric for main chamber mixtures close to stoichiometry, the vaporisation of the water reduces the heat losses and increases the piston work by steam expansion. The load is controlled by changing the amount of H_2 introduced in the main chamber and the premixed-to-diffusion combustion ratio. The H_2O injection only operates at high loads. The preliminary evaluations of this combustion system have shown the opportunity to burn mixtures from very lean to stoichiometric with very high efficiencies and using the water injection for steam expansion and temperature control.

Options being presently considered include the use of a second main chamber direct hydrogen injector to provide fast delivery of significant amount of the low density fuel in large bores as well as the use of multiple main chamber direct water injectors.

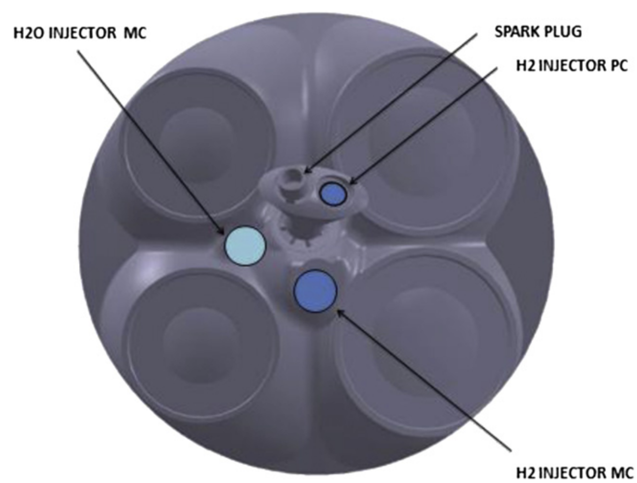

Fig. 1 – Direct injection jet ignition combustion system.

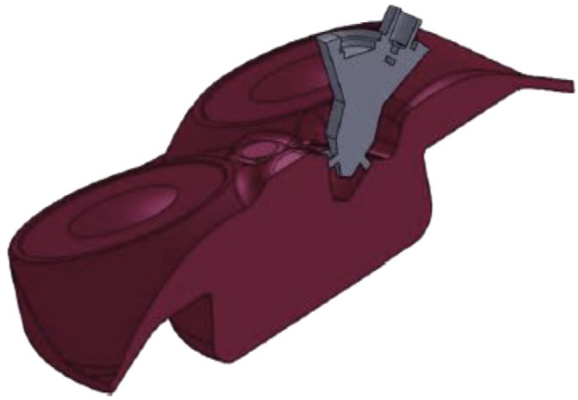

Fig. 2 – Sketch of the combustion chamber and the jet ignition pre-chamber of a direct injection spark ignition gasoline engine gasoline engine modified to run hydrogen direct injection and jet ignition (from [4]).

The steps taken in the quest for greater thermal efficiency are the bulk combustion with shaping of the heat release and the heat transfer rates through the shaping of the fuel and water injections and the phasing of the ignition event. The coolant, exhaust and air charger heat recovery discussed later further increases the opportunities to achieve significant improvements of the fuel economy.

3. Novel waste heat recovery systems

The increasing interest in emission and fuel consumption reductions calls for measures that ensure the optimal utilisation of the fuel energy. The internal combustion engine exhaust gas energy is by far the most attractive among the waste heat sources because of the heat flow and the high temperature. In a turbocharged, direct injection engine with air cooler, the top fuel conversion efficiency are about 45%, and the exhaust gas energy accounts for more than one half of the waste heat, approaching 25–30% of the fuel energy. About one half of the remaining 25–30% is then lost in the air cooler, and similarly in the water and oil coolants, with the heat radiation contributing about 1–5% of the fuel energy. Changing the load and the speed, the percentage of fuel energy lost in the waste heat sources increases significantly even in throttle-less engines controlled by the quantity of the fuel injected (at idle, theoretically the 100% of the fuel energy is lost, being the brake efficiency defined as the power at the crankshaft divided by the product of fuel flow rate and lower heating value), and the proportions of the waste heat sources also change, with the coolants considerably increasing their weight. The internal combustion engine coolants energy is less attractive than the exhaust gas energy because of the low temperature, but still relevant for the significant heat flow. It is possible to generate an additional mechanical output of 10% the main engine power at top fuel conversion efficiency operation by utilising the exhaust and coolants energy in a waste heat recovery system comprising both steam and power turbines, and even larger percentages working off top efficiency operating points [16].

Fig. 3 presents the WHRS with gas and steam turbine. Water moves through the coolant engine passages (WHR1) driven by the water pump working as a pre-heater of the fluid collecting engine coolant and oil coolant heat. The water then enters the cooler of the compressed air (WHR2) to further approach the boiling conditions collecting the air cooler heat. The water/steam mixture finally enters the heat exchanger on the exhaust, immediately downstream of the turbocharger turbine, the de-NOx catalyst and the power turbine (WHR3) to produced superheated steam. The superheated steam then expands in a steam turbine (ST1) and then enters the air-cooled condenser (WHR4). The steam turbine and the condenser are by-passed during the transient phase for a much quicker warm-up of the engine during cold start driving. The power turbine (GT2) is disengaged and by-passed when not convenient for the overall efficiency of the engine.

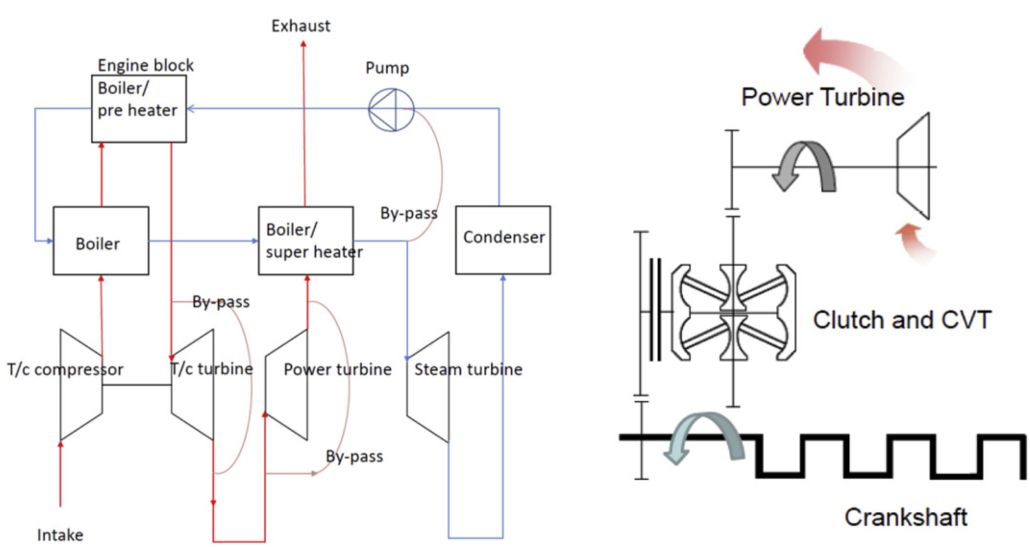

Fig. 3 – Waste heat recovery system with gas and steam turbine and sketch of the power (and steam) turbine connection to the crankshaft.

The power turbine as well as the steam turbine and the pump are connected through individual CVT to the crankshaft. This permits a fully mechanical system. By-pass operates on the steam as well as the power turbine (GT2). A by-pass may also be necessary on the turbocharger turbine, but is not presently considered.

4. Computational results

Preliminary results have been obtained with methane rather than hydrogen. This is because some experimental data were available for a similar heavy duty truck engine converted to operated dual fuel with Diesel pilot and main LNG (liquefied natural gas injection) to tune the combustion model parameters otherwise very difficult to be determined. The water injection is also disabled.

The main data of the 12.8 L HDT Diesel engine are presented in Table 1. The baseline Diesel engine without WHRS has a target performance of 2600 Nm torque 1000–1450 rpm, and of about 400 kW of power 1450–1900 rpm, with BSFC values around 190 g/kW h, corresponding to brake fuel conversion efficiencies of 44%. The baseline Diesel engine is compliant with EURO-5 emission standards.

While the previous papers have analyzed the combustion system with non-predictive combustion models set up on the basis of the Diesel only experiments, this paper also presents results of simulations with the non-predictive combustion models further tuned on the basis of predictive combustion model results.

Figs. 4–6 present the computed full load power and torque curves and the brake efficiency and operational fuel to air equivalence ratio λ maps.

The brake efficiency is crankshaft power to product of fuel flow rate by fuel lower heating value. The top efficiency is above 50% with λ about 1.3. The efficiency is very high up to one third of the load over the full speed range.

Table 1 – 12.8 L in-line six cylinder turbocharged directly injected diesel engine.

Displacement per cylinder [1]	2.13
Number of cylinders	6
Engine layout	I-6
Compression ratio	18
Bore [mm]	131
Stroke [mm]	158
Connecting rod length [mm]	300
Wrist pin offset [mm]	0
Clearance volume [1]	0.125
Engine type	C.I.
Number of intake valve per cylinder	2
Intake valve diameter [mm]	46.3
Intake valve maximum lift [mm]	12.1
Number of exhaust valve per cylinder	2
IVO [deg]	314 (−46)
IVC [deg]	602 (+62)
Exhaust valve diameter [mm]	41.5
Exhaust valve maximum lift [mm]	13.4
EVO [deg]	100 (−80)
EVC [deg]	400 (+40)

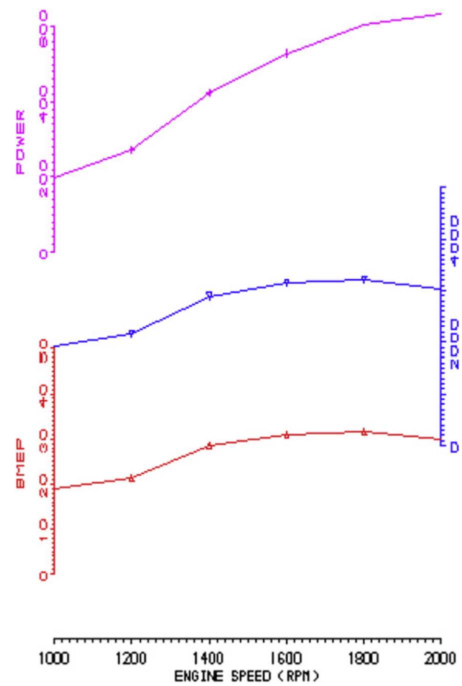

Fig. 4 – Brake torque and power curves – CNG engine – single fuel.

These results have been obtained with ICE and WHR system GT-SUITE models [12]. GT-SUITE simulations are a very well-known procedure. The author group has published extensively on the GT-SUITE model of the engine and the waste heat recovery system. The Wiebe functions used to model the combustion evolution have been computed first from the diesel only experiments then corrected on the basis of CFD results [13–15]. The heat release profile is an input to the GT-SUITE model, being the software unable to tackle the modelling of the complex phenomena involved in the new combustion systems. Obviously these computational results do not have the same validity of experiments with a prototype engine, and the sources of inaccuracy may be significant in such a non-conventional application.

For what concerns the WHR, the ICE model provides for every operating point BMEP and speed the heat flows in the engine coolant passages, the coolant and the exhaust heat exchangers. These quantities are used to set up the boundary conditions for the WHR model. The WHR model results then

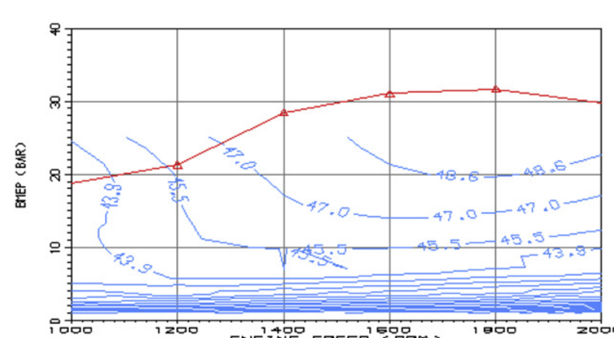

Fig. 5 – Brake efficiency map – CNG engine – single fuel.

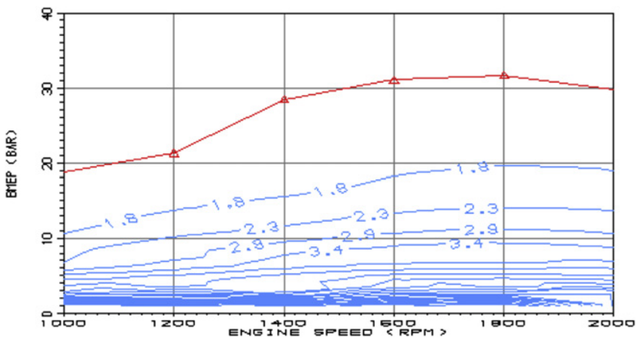

Fig. 6 – λ map – CNG engine – single fuel.

provide some of the boundary conditions for the ICE model. Because the WHR and the ICE boundary conditions are dependent each other, some iterations are needed to reach results in reasonable agreement between the two uncoupled simulations. A better solution is obviously the development of an integrated GT-SUITE model for the ICE and the WHR as presently being considered.

The results are much better than what has been proposed for the transportation engines designed so far, and close to the results obtained for large stationary engines for power generation or engines for marine applications [16–18]. These latter engines run much lower speed and have some advantages in terms of heat release (but obviously some disadvantages in terms of heat transfer) affecting the indicated mean effective pressure but much lower friction mean effective pressure for the low mean piston speed. The proposed solution has a more flexible combustion and heat recovery systems.

In the single fuel jet ignition simulations, the pre-chamber volume is 5% of the combustion chamber volume and it is fuelled with a locally stoichiometric amount of methane. All the results were obtained mostly with a mixed combustion mode. Part of the methane is injected in the bowl prior of the jet ignition, the remaining concurrent and post. Premixed and diffusion amounts change with load and speed and with the diesel or the jet ignition design. The concept is still very far from being optimised.

5. Conclusion

The paper has presented the latest concepts for combustion and waste heat recovery systems being considered for hydrogen engines.

The combustion system uses high compression ratio, bowl-in-piston combustion chambers with direct injection of the hydrogen. Combustion is started by jet ignition. The temperature is controlled by water injection also providing steam expansion. The waste heat recovery system uses gas power and steam turbines. Preliminary simulations show improvement of top fuel conversion efficiencies to above 50% also in the high power density operation.

Faster warm-up during cold start is also possible reducing the cold start fuel penalty of driving cycles. This H_2ICE with a fully mechanical WHRS is ideally coupled to a mechanical kinetic energy recovery system for fuel energy efficiency better than electric cars.

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